Report No. ZR-659-053

TAPERED BOLTS--THEIR INFLUENCE ON FATIGUE OF AIRPLANE STRUCTURES

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C.R. Smith

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CONTENTS

ILLUSTRATIONS

SUMMARY:

Fatigue data are presented showing the effects of interference fit bolts or bushings on the fatigue life of structures. It was shown that fatigue lives were improved thirty times by using proper amounts of interference. A theory is presented for estimating amounts of interference required for optimum fatigue life.

INTRODUCTION:

This report is a continuation of the work presented in Ref. 1 on Fatigue Resistant Structures. In that report, the author presented data showing effects of overloading on the fatigue strength of riveted joints.

Here, an attempt will be made to make use of the linear strain theory in combination with experimental effective spring constants for predicting effects of pressed fit bushings on lugs and interference fits on bolted joints.

THEORY:

In Reference 1, the author showed that small amounts of static overloading was detrimental to the subsequent fatigue life of riveted joints, while high overloading improved life. Test data for one of these joints, a lap joint of 0.051 - inch-thick clad 7075-T6 sheet, using 3/16 100° countersunk rivets spaced 1 in. apart in three rows ½ in. apart, are given in Table I. It will be noted that where prestresses of 18,000 psi were applied prior to fatigue testing, a substantial reduction in life was had--especially at the 10,000 psi and 5,000 psi levels. Preloads of 25,000 psi caused some reduction in life, while preloads of 42,000 psi caused marked improvements in life. The loss and subsequent improvements were explained as losses in propping* action afforded by the rivet, and to gains in residual compressive stress and re-distribution of load between rivets. The riveted joint in itself is too complicated to use as a vehicle to illustrate this mechanism. Accordingly, an attempt will be made here to analyze the same mechanism, using a simple lug with bushings pressed in at various amounts of interferences.

In Table II are presented data on the effect of pressed-in hardened steel bushings on fatigue life of a 7075-T6 lug which was 3.750 in. wide and 0.75 in. thick.(2) The hole diameter was 1.250 in.,

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* It was assumed that tightly driven rivets would act like props which would prevent the return of stress to zero, even when unloaded. Slight amounts of overload would tend to loosen the rivet, so that it could not provide the support needed for this effect.

affording a theoretical etrese concentration of 3.5. Bushings of hardened steel having wall thicknessee of 1/4 in. and 1/8 in. were preesed in at varying amounts of interference, from zero to 0.010 in. Bushings were chamfered 10° to enable preeeing in without extruding material from the lugs. Chamfcred ends were ground off after pressing.

It will be noted that the average failing life for four epecimens having 1/8 in. bushings, but no interference, wae 29,250 cycles for repeated nominal etresses of 18,800 psi (net). With 0.0075 in. interference the average life was increasod to 1,089,000 cycles. Average livee were less for either greater or lesser amounte of interference. Theee data are plotted in Figure 1.

In Figure 2 are are S-N curves for smooth axially loaded 0.2 in. diameter epecimens of 7075-T6 aluminum alloy as presented by Howell and Miller[3] at the 1955 ASTM Meeting. In Figure 3 are the same curves[4] over which a line hae been drawn to represent the locus of fatigue livee for varying etrese ratioe wherein a peak strees of 66,000 psi was maintained. In this case, 66,000 psi represents the nominal stress times the theoretical etreee concentration factor for the lugs. Two test points are plotted on thie line, one for the life of a lug with no interference and the other for a bushing interference of 0.0075 in.

Here, it can be seen that the fatigue life for the lug without interference falls about on the curve for R = 0 (actual teet R was 0.015) indicating the theoretical concentration factor of 3.5 was about right without further correction* The point representing

* Fatigue reduction factors or Neuber values are in common usage. It is possible that the bolt bending effect, which was not considered here, was enough to offset these effects.

0.0075 in. interference is open to question at this time, because it would appear that the 0.0075 in. interference should add to the peak stress. However, if we consider that the 0.0075 in. interference amounted to a nominal strain of 0.006 in./in. (assuming E = 10,000,000 psi. The hole in the lug was 1.25 in. diameter) we find that an additional 0.0006 in./in. strain would have been required to increase the maximum stress. It is assumed that the strain due to press fit at locations away from the concentration did not contribute to the fatigue damage.

According to the position of the point representing the lugs with 0.0075 in. interference (average for four specimens), and the relationship between the S-N curves for various stress ratios, it would appear that an effective stress ratio of about 0.6 would be in order. The term effective is used here because the lug behaved as though that were the stress ratio, despite the fact that actual loading was at a stress ratio of 0.015. Since the definition of stress ratio is the ratio of the minimum stress divided by the maximum stress, the minimum stress would amount to 66,000 psi x 0.6 = 40,000 psi. Assuming a modulus of elasticity of 10,000,000 psi for aluminum alloy, the strain for 40,000 psi would have amounted to 0.004 in./in. In other words, the 0.0075 in. interference resulted in expanding the hole 0.004 in./in., while the bushing shrunk 0.002 in./in. (remember the 0.0075 in. interference amounted to 0.006 in. interference per inch of diameter.)

We can now say that the effective* spring constant of the bushing was twice that of the lug material, and, knowing the relative spring constants, we can now predict fatigue life for any other amount of bushing interference. This can be best illustrated by a numerical example, however before tackling this problem, a schematic of what we've been discussing should prove of assistance.

A mechanism to explain these phenomena is best had by an analogy. This time it has to do with springs. Remember, the assumption was made that the bushing acted as a prop, adding a floor, so to speak, under the stress pattern which prevented it from returning to zero.

In Figure 4 are schematic sketches of a spring subjected to repeated tension loads.[4] Figure 4a shows the sine curve generated by an unrestrained spring for a given loading. The spring here is a simulation of the most highly stressed inside fibers of a lug without bushing. Figures 4b, 4c, and 4d show the spring subjected to the same loading, but restrained by posts of varying heights, simulating bushings with different amounts of interference. It will be noted that the stress amplitude in these sketches represent a movement of both, lug and bushing, however, in Figure 4b, the bushing deformation is relatively small compared to that for the spring representing the lug. This simulates the case where only a small amount of interference was present.

* The term effective is used here, since prorating the moduli of elasticity and areas of the bushing and lug would result in nothing like this ratio. This would lead one to believe that only the areas adjacent to the faying surface contributed much to the stiffness, e.g., were the ratio of spring constants 3 to 1 instead of 2 to 1, this would be the case. Accordingly, it would seem that only a portion of the lug cross section area was engaged in reacting the forces introduced by the bushing.

Figure 4c shows a case where the interference was almost equal to the cyclic stress. The relative stiffnees is evident from the small amplitude, the amplitude here being representative of the combined spring constants of both lug and bushing. This represents the optimum amount of interference to be had for this particular stress level. Greater amounte of interferenco such as shown in Figure 4d add to the peak stress so that the fatigue life could be expected to be less than for that shown in Figure 8c*, although it could easily be greater than for that shown in Figures 8a and 8b by virtue of the emall cyclic amplitude.

It should be pointed out that one and only one amount of interference would be optimum for a given etress amplitude. While that shown in Figure 8c would be optimum for the amplitudo shown, higher loading would benefit more by greater interference such as shown in Figure 4d, while for lower loading, the interference shown in Figure 4b might be best.

An attempt will now be made to apply the above spring analogy directly to etrain, using the experimental data from Table II for the neceseary spring constants, and S-N data for emooth axially-loaded specimens for equivalent lives.

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* The cyclic amplitude, once the combined spring constants are fully employed, would be the same for any amount of interference greater than shown in Figure 4c. Photoelastic studies by Jessop, Snell and Holister lend support to this supposition. [3]

We have shown that an interference of 0.0075 in. would induce a residual stress in the lug of 40,000 psi. Assuming that the combination bushing and lug acts as a single spring and that the rate is constant from zero to 40,000 psi, were the bushing to be pressed in with an interference of 0.005 in., the residual stress would be 0.005/0.0075 x 40,000 psi = 26,600 psi. The new stress ratio would be 26,600/66,000 = 0.4 and the predicted fatigue life from the S-N curves given in Figure 2 would be about 300,000 cycles. This is in good agreement with the experimental average values of 316,750 for the 1/4 in. thick bushings and 255,250 for the 1/8 in. thick bushings.

It is of interest to note that little difference was found between fatigue lives for lugs with 1/8 in. thick or 1/4 in. thick bushings. This may be attributed to stiffness ratios being effective only in the faying surfaces of the journal, or to differences in bolt bending, bolts being 0.75 in. and 1 in. diameter for the 1/4 and 1/8 in. thick bushings, respectively.

So far our discussion has covered only those conditions where interference did not add to the cyclic stress. In the case of the bushing having an 0.010 in. interference, some addition to the peak stress could be expected because this would amount to an equivalent interference of 0.010/1.25 or 0.008 in./in. This would be higher than the 0.0066 in./in. peak cyclic strain assumed in these experiments.

A quantitative estimate of how much this would add to the peak stress can be had by adding the cyclic stress amplitude to the residual stress set up by interference. First, the residual stress would amount to just twice that found for the 0.005 in. interference or 2 x 26,600 psi = 53,400 psi. It is assumed that the spring rate is linear throughout these ranges. Secondly, the cyclic amplitude for the unbushed lug was found to be 66,000 psi at the point of stress concentration. However, the spring constant of the combined bushing and lug at this point is now three times* that of the unbushed lug, so the cyclic amplitude would be one-third as much or 22,000 psi. Accordingly, the stress for 0.010 interference would amount to 53,400 psi residual stress plus 22,000 psi clic stress, giving a total of 75,400 psi. The corresponding stress ratio would be 53,400/75,400 = 0.71 and the predicted life from Figure 2 would be about 700,000 cycles. The test average for the 1/4 in. thick bushings was 935,000 cycles (average of 4 specimens).

Static preloading above the elastic limit introduces a problem that is relatively easy where no interferences are present, but become very complex with interference. Especially pertinent are the compressive residual stresses acquired at the points of concentration. A method for determining the amount of residual stress,

* It was previously shown that the bushing was effectively twice as stiff as the high stressed region of the lug. Accordingly, the combined spring constants would be the sum of both or three times that of the original lug. This is true whether or not both springs (lug and bushing) act in the same direction.

employing a linear strain relationship in the plastic range, has been used by the author with satisfactory results for open notches.[4] In essence, it assumes that a concentration is a geometric relationship and would be more appropriately expressed in terms of strain than stress. Accordingly, in order to find the stress at the point of concentration, one only has to multiply the nominal strain times the concentration for maximum strain, using stress values from ordinary stress-strain curves.

Since only a very small portion of the cross section area suffers plastic deformation, this material will be forced into compression on unloading. The amount of compression will depend largely on the amount of plastic deformation, however, for small amounts, it can be assumed to be equal to the permanent set (strain between loading and unloading lines) times the modulus of elasticity. Where large amounts of plastic strains are involved, residual stresses equal to or greater than the yield stress of the material obviously could not be held. Accordingly, an arbitrary value of 2/3 the yield stress has proven to be a reasonable assumption.

For estimating fatigue life after a load causing plastic deformation at the point of stress concentration, it is an easy matter to translate the whole stress program downward by the amount of residual stress. For example, if this residual stress amounted to 33,000 psi in compression, the lug which formerly experienced 0 to 66,000 psi stress cycling, would now be subjected to ± 33,000 psi. Note that the stress range is the same in both cases, the manipulation here being only a translation downward of the mean stress. Similarly, were the residual stress -22,000 psi, the corresponding cyclic stress would be from -22,000 psi to +44,000 psi for the stress range of 66,000 psi. Also, if the residual stress were -44,000 psi, the maximum stress would be + 22,000 psi, and the new stress ratio would be -2 (R = -2).

We can now use the S-N curves shown in Figure 2 to determine the fatigue life of our original lug for the various amounts of residual stress assumed in the last paragraph. These points are plotted in Figure 5. Note the orderly fashion in which fatigue life seems to increase with lowering of mean stress (increase of residual stress). It should be remembered that these points represent the life of the same lug for the same loading condition, the only difference being the previous stress history; e.g., it was assumed that the lug were overloaded sufficiently to acquire residual compressive stresses in the above amounts.

We have shown that an increase in fatigue life can be expected in lugs having pressed in bushings; also that an increase can be expected with static overloading prior to testing. However, overloading a lug having a pressed in bushing would tend to loosen the bushing which would shorten fatigue life, while at the same time acquiring beneficial residual compressive stresses which would lengthen life.

Whether a net gain or loss could be expected, would depend largely on the configuration, amount of bushing interference, and size of preload. In no case, would the life be expected to be less than predicted by a plot such as shown in Figure 5. Note that this particular plot is for a stress range of 66,000 psi, however, similar plots could be drawn for any stress range..

One set of lugs having an 0.0075 in. interference was prestressed to 48,000 psi prior to cycling at 18,800 psi nominal stress. The average failing value was 541,000 cycles (average of four specimens). Assuming that this overload was sufficient to relieve all of the interference, it appears as though the stress ratio was in the neighborhood of -2.2, and the residual stress would had to have been about -44,500 psi.* This point is plotted on the graph in Figure 5.

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* According to the assumptions made in the linear strain theory, a 48,000 psi preload should have induced a residual stress higher than the yield strength of the material. Therefore, a further assumption was made that only 2/3 of this amount could be sustained, and a calculated residual stress would have been 2/3 x -70,000 psi = -46,600 psi instead of the -44,500 psi shown above. These differences, however, would not have materially changed the predicted fatigue life.

Unfortunately, no other experiments were made on static overloading of lugs with pressed in bushings. The linear strain theory would predict that a 38,000 psi prestress would have been sufficient to relieve all of the bushing interference, whereupon the subsequent life would have been 400,000 cycles. This would represent the greatest loss to be expected for this particular configuration. Greater amounts of overloading would increase life by acquiring additional beneficial residual compressive stresses, and lesser amounts would not completely relieve benefits of pressed in bushings. It should be noted, however, that 400,000 cycles is substantially greater than the life for a lug without bushing.

Further experiments were made on small lugs, using taper pins[5] instead of pressed in bushings. These data are presented in Table III and plotted in Figure 6. It will be noted that roughly the same amount of fatigue improvement was had here as with the pressed in bushings, lending further support to the idea that the added stiffness is effective only in the faying surfaces (in the journal), e.g., the 1/8 in. thick bushings, 1/4 in. thick bushings and solid taper pin resulted in about the same amount of fatigue improvement.

Still further experiments were made on small lap joints using tapered bolts. In some cases, the holes in the doubler at the first fasteners were drilled oversize in order to force the center fastener to carry more of the load. These are described under Experiment C.

EXPERIMENTS:

A. EFFECTS OF PRESS FIT BUSHINGS ON FATIGUE LIFE OF LUGS

The purpose of this experiment was to determine the effect of interference fit bushings on the fatigue life of 7075-T6 lugs having a theoretical stress concentration of 3.5.

Four specimens each were prepared of 0.75 thick 7075-T6 as shown in Table II. They were subjected to repeated loading of 18,800 psi (R =.015) in a Sonntag SF-10U axial loading fatigue testing machine.

Hardened steel bushings (HT 150,000 - 180,000 psi) were prepared with 10° chamfered ends to permit pressing in the lugs without extruding material from the lugs. Chamfered ends were machined off after pressing. Bushing wall thicknesses and interferences are given in Table II.

Table II shows average fatigue lives of lugs together with standard deviation values for individual lots. These data are plotted in Figure 1, where a graph showing effects of various amounts of interference versus fatigue life is presented.

One lug, having a bushing interference of 0.0075 in., was statically loaded to 48,000 psi prior to fatigue testing. It had approximately one half the life of similar lugs that were not prestressed.

EXPERIMENTS: (Contd)

B. TAPER PINS - FATIGUE LIFE VERSUS INTERFERENCE

These experiments were made on 0.1 thick by 1 in. wide 7075-T6 lugs having an 0.375 in. diameter hole 3/4 in. from one end. Various amounts of interference were obtained by drawing up taper-pins to the desired values. The pins, having a taper of 1/4 in. per foot, provided an interference of 0.0083 in. per turn on the nut. Each specimen had a zero reference point corresponding to a position attained by loading the butt end of the taper-pin with a 15 lb weight. Holes in the fixturee were enlarged to permit drawing up the pins without binding against the sides. All tests were made in a Sonntag SF-1U axial loading fatigue testing machine at a etress ratio of 0.1 (R = 0.1).

Fatigue data are presented in Table III and a graph of interference values versus fatigue life is given in Figure 6. Graphs comparing fatigue lives of lugs with 0.0035 in. interference and those with no interference are presented in Figure 7. The strees values shown are for gross section away from the hole. It should be noted that stresses for lugs are usually presented in terms of nominal net areas; this was an expedient, however, for correlating data obtained in this manner with nominal stress values normally used in airplane stress analysis. Nominal net stresses can be had by dividing values shown by 5/8.

EXPERIMENTS: (cont.)

C. TAPER-LOK BOLTS IN CLAD 7075-T6 SHEET

In view of the success with taper pins having interference fits, these experiments were made using Taper-Lok bolts. Taper-Lok is a trade name for a bolt having 1/4 in. taper per foot, a product of the Briles Manufacturing Co., 1415 Grand Ave., El Segundo, Calif. These bolts are matched with special drill-reamers, so that control of interference can be had by measuring the head protrusion above the sheet prior to tightening the nut. An 0.003 in. interference was had in these tests by setting the bolt heads 0.15 in. high before drawing up, stops on the drill-reamer being adjusted by trial in sample test pieces.

The control specimens consisted of 1 inch wide lap joints of Clad 7075-T6 0.10 in. thick. They were fastened with four 100° 1/4 inch diameter countersunk rivets spaced one inch apart. An edge distance of 1/2 inch was kept. The rivets were oriented so that the countersunk heads of the first two rivets on each sheet were towards the loaded end. Thus, a failure in any specimen represented the lowest life for two. All specimens were cycled at 10,000 psi gross stress (R = 0.1) in a Sonntag SF-1U axial loading fatigue testing machine.

The modified specimens were the same as the controls, except that the end rivets were replaced with Taper-Lok bolts drawn up to provide an interference of 0.003 in. In the Lot 1 specimens, EXPERIMENTS: (cont.)

this interference was held through both, doubler and top sheet. In Lot 2 specimens, the holes in the doubler were drilled 0.064 in. oversize so as to make the first fastener in the joint incapable of holding a shear load, other than through clampling; holes in the top sheet, however, were held 0.003 in. undersize. The results of these tests follow:

FATIGUE OF CLAD 7075-T6 LAP JOINTS WITH TAPER-LOK BOLTS

Life—Cycles to Failure for Repeated Stress of 10,000 psi

	Riveted Control	Taper-Lok Bolts with 0.003 Interference Lot 1 *	Lot 2 **
	233,000	434,000	1,677,000
	76,000	397,000	1,433,000
	96,000	247,000	1,003,000
	128,000	267,000	
	52,000	271,000	
Average	117,000	323,000	1,371,000

* Same as controls, except first rivets replaced with Taper-Lok bolts having 0.003 in. interference in countersunk sheet and doubler.

** Same as controls, except first rivets replaced with Taper-Lok bolts having 0.003 in. interference in countersunk sheet with 0.032 clearance on opposite.

EXPERIMENTS (cont.)

D. TAPER-LOK BOLTS IN BARE 7075-T6 SHEET

Specimens and tests for these experiments were made in the same manner as for those in Experiment C with the exception of using bare instead of clad material, and omitting the Lot 1 type specimens.

S-N data were developed from approximately 10^4 to 10^7 cycles. These are given in Table IV. S-N curves comparing fatigue lives for riveted control specimens and those with Taper-Lok bolts are shown in Figure 8.

E. STATIC TESTS ON JOINTS HAVING OVERSIZE HOLES IN DOUBLERS

Two types of specimens were used, one having four tandem fasteners and the other with two. As in experiments C and D, first fasteners towards the loaded end were reverse oriented to cause the critical stress to occur at the high loaded fastener. Instead of using 1 inch wide sheets as in Experiments C and D, the widths were increased to 1.5 inches to make the joints critical in bearing instead of tension.

They were tested in a Southwark-Emery hydraulic testing machine, using a dial indicator extensometer which spanned the joint for measuring deformation and set. The yield strength was chosen as a permanent set of 2½% of the nominal rivet diameter or 0.00625 in. to conform with data in ANC-5 for 1/4 in. diameter rivets.

EXPERIMENTS (cont.)

The results of these tests are presented in Table V, values given being the ratios of the strengths for joints with oversize holes at the first fastener divided by that for a conventional joint. Each value represents an average for four specimens.

It is to be noted that each joint having an oversize hole at the first fastener represents a removal of 50% bearing area insofar as yield strength is concerned. Accordingly, the prorated yield strengths for the data in Table V would amount to 0.50 for the solid fasteners and remainder for those with oversize holes. Thus, for the 2 rivot joint of 0.064 sheet (lowest of all test values), the yield strength for the fastener with oversize hole would amount to 0.76 - 0.50 or 0.26. In terms of strength per fastener, this would amount to 0.26/0.50 or 52% of its original strength.

It is assumed that no loss in bearing strength was had for ultimate loading conditions. The ultimate load values in Table V confirm this.

DISCUSSION:

In comparing the data from the pressed in bushings with those for taper pins, it will be noted that an apparently higher etrain interference was required for the taper pins than for the pressed in bushings for about the same amount of improvement in fatigue life. This may have been due to experimental inaccuracy in determining the zero interference reference point, which would have been more critical in a 3/8 in. dimension than for 1¼ in. This also may be an explanation for the relatively large standard deviation values where interference was present ae compared with the controls. However, the life corresponding to 2 standard deviations less than the log mean (97½% of all specimens should exceed thie value) for specimens having 0.0035 interference was at least four times the equivalent value for the controls.

It ehould be noted that all of the tests made in these experiments were at room temperature. In the case of elevated temperatures, it is quite probable that some of the residual tension streeses set up by bolt or bushing interference would disappear with continued operation through relaxation.

Elevated temperatures would not only be instrumental in relaxing residual tension stresses, but compreesive as well. In the case of all welded structures, improvements in fatigue life,caused by upper level loading,may be erased by losses in residual compressive stresses by relaxation. Specifically, this would mean that the fatigue life,as determined by spectrum type tests at elevated temperature, DISCUSSION: (cont.)

would be unconservative unless long waiting periods at elevated temperature were had after application of the highest load in the spectrum before continuing at lower level loading. With tests designed to have at least 50 sequences before failure (50 times through the complete spectrum of loads and temperatures), this would necessitate an unacceptably long test period.

While these effects would be almost impossible to predict without long-time temperature exposure and slow cycling tests, it is possible that a reasonable approximation could be had by extrapolating a few data from short-time temperature and load exposure tests, using ordinary material creep data as a guide. This, however, could not be accepted without considerable research on small notched specimens over long periods of time.

It should also be noted that certain materials may become more vulnerable to corrosion when subjected to tensile stresses, commonly called stress corrosion. This is especially true when the tension stress is in the short transverse direction. Accordingly, where interference fits are used, reasonable precautions should be taken for adequate protection against exposure to corrosive elements.

CONCLUSIONS:

A method for determining the effect of interference fits in joints employing mechanical fasteners has been presented. While not altogether rigorous, a good agreement with test data was had. More work needs to be done regarding fatigue loss with overloads, especially towards obtaining structures incapable of inducing loads causing these losses. Also, it is apparent in cases where elevated temperatures are involved, that some loss in residual stresses could be expected. Whereas the residual tension stresses in themselves are not beneficial fatiguewise, it is through their influence that the increase in spring constant afforded by the bolt or bushings is made available.

A mechanism such as described, or one very similar, should lend itself for estimating cumulative damage in fatigue, however, more work would be required to obtain necessary spring constant ratios of rivets or other fasteners to their surrounding structures.

When used with elevated temperatures, an additional time-relaxation correction factor would be required. These factors, however, would have to be determined experimentally.

REFERENCES

1. C.R. Smith, "Fatigue Resistant Structures", Convair Report No. ZR-658-030

2. D.H. Love, "Lug Design - Bushings - Interference Fit - Fatigue Tests", Convair Report No. 9486

3. F.Howell and J.L. Miller, "Axial-Stress Fatigue Strengths of Several Structural Aluminum Alloys", Proceedings of the American Society for Testing Materials, 1958, Table II.

4. C.R. Smith, "Design Applications for Improving Fatigue Resistance of Airplane Structures", Advance Copy of paper presented at the A.S.T.M. June, 1960 Meeting held in Atlantic City

5. C.R. Smith, "Fatigue Resistance - Design Considerations", Aircraft Engineering, May 1960

TABLE I

Effect of Prestress on Life of Riveted Lap Joints

Maximum Cyclic Stress (Psi gross) R = 0.1	Maximum Static Prestress 0	18,000	25,000	42,000
25,000	4,000 cycles** 4,000 0.1077			20,000** 20,000 0.1692
18,000	15,000 15,000 0.1068		13,000 13,000 0.0575	155,000 133,000 0.3010
15,000	36,000 34,000 0.1712	38,000 32,000 0.2720	127,000 81,000 .512	812,000 748,000 0.2130
10,000	355,000 350,000 0.0812	102,000 98,000 0.1364	175,000 167,000 0.1494	10,169,000*
7,500	1,152,000 1,140,000 0.0944			
5,000	6,752,000*	1,466,000 1,462,000 0.0316	4,940,000*	
3,500		9,818,000*		

* One test point only

** Values in this table are for Arithmetic Mean and Log Mean Lives, and Standard Deviation, respectively. Standard Deviations are with respect to log mean lives.

TABLE II

Effect of Pressed-in Bushings on Fatigue Life of 3/4 in. Thick 7075-T6 Lugs having K_t = 3.5, and Hole Diameter = 1.25 in.

Bushing Wall Thickness	Nominal Interference	Interference per inch of Hole Diameter	Average Life* (4 Specimens)	Standard Deviation
No bushing	--1¼ in. dia.bolt		34,500	0.095
1/8 in.	0.000	0.000	29,250	0.165
1/8 in.	0.005 in.	0.004	255,250	0.284
1/8 in.	0.0075 in.	0.006	1,088,750	0.193
1/8 in.	0.0075 in. (48,000 prestress)	0.006	541,250	0.403
1/4	0.005 in.	0.004	316,750	0.269
1/4	0.0075	0.006	990,000	0.212
1/4	0.010	0.008	935,000	0.166

* Loaded at repeated 18,800 psi nominal net stress, R = 0.015. All tests made on Sonntag SF 10U Axial Loading Fatigue Testing Machine.

TABLE III

Effect of Interference on Fatigue Life of Pin Loaded Lugs

Stress (ksi gross) R = 0.1	Log Mean Life and Standard Deviation			
	Control-- No Interference		With 0.0035 in. Interference	
	Log Mean*	Std. Deviation	Log Mean*	Std. Deviation
21,000	----	----	50,900	0.443
18,000	13,500	0.0894	316,300	0.402
15,000	21,400	0.0914	474,600	0.452
12,000	34,500	0.0522	894,000	0.116
9,000	46,000	0.0117	1,396,000	0.168
6,000	174,000	0.3922		
4,000	953,000	0.2571		

* Log mean life for three specimens. Each specimen was double ended, so that each value represents the lowest of two. Standard deviations, however, were based on three specimens.

TABLE IV

Effect of Interference Fit on Fatigue Life of 7075-T6 Lap Joints

Specimen Type	Stress Level (ksi) R = 0.1	Log Mean	Log Mean Life (Cycles)	Standard Deviation
	30	3.301	2,000	0.224
4 Rivet	20	3.992	9,900	0.0906
Control*	15	4.322	21,000	0.0245
	10	4.765	58,000	0.2030
	8	----	129,000 (1 specimen)	
	6	5.492	310,000	0.1230
	4	6.916	8,240,000	0.0283
2 Rivets	30	4.238	17,300	0.1375
in Center with	25	4.452	28,300	0.0756
Taper-Lok Bolts	20	4.812	64,800	0.0173
at Ends--0.002	15	5.116	131,000	0.1170
Interference on	10	5.533	341,000	0.1150
Csk. Side & 0.032	8	-----	632,000 (1 specimen)	
Clearance Opposite	6	over 10,000,000, test stopped		

* 1 inch wide lap joints having four 1/4 diameter countersunk rivets in tandem spaced @ 1 inch. Rivets were oriented so that countersunk heads of first two rivets on each sheet were towards loaded end.

TABLE V

TAPER-LOK BOLTS - RATIO OF STATIC STRENGTHS FOR
JOINTS WITH OVERSIZE HOLES IN DOUBLERS TO RIVETED CONTROLS

No. of Fasteners in Specimen*	Sheet Thickness and strength ratios**									
	0.064		0.072		0.080		0.090		0.100	
	yield	ult.	yield	ult.	yield	ult.	yield	ult.	yield	ult.
4	0.96	1.00	0.96	0.99	0.94	0.99	0.85	1.00	0.83	1.03
2	0.76	0.99	0.80	1.01	0.86	1.04	0.93	1.08	0.86	1.12

* Specimens were 1-½ in. wide. In the riveted controls, all fasteners (rivets) were driven tightly, the end rivets being oriented in opposite directions so that the countersunk side was nearest loaded end. Taper-lok bolted specimens were prepared in a like manner --e.g., countersunk head (0.003 interference) was nearest load, while opposite sheet had 0.032 in. clearance. The 4 fastener joint had 2 tightly driven rivets in center. 2 fastener joint had 1 tightly driven rivet.

** Strength of joint with oversize holes/conventional joint

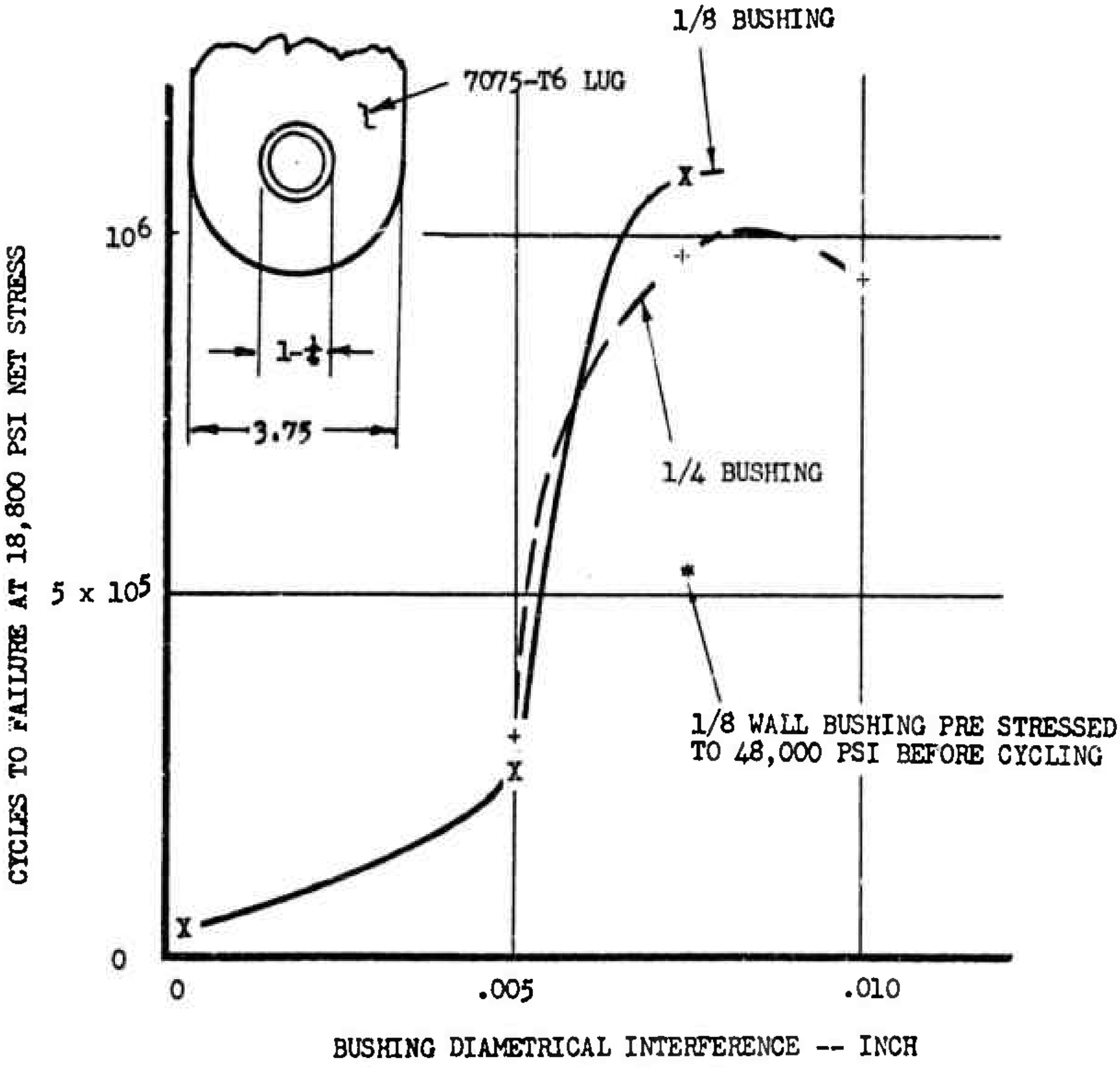
1/8 BUSHING
7075-T6 LUG
1-¼
3.75
10^6
5×10^5
0
CYCLES TO FAILURE AT 18,800 PSI NET STRESS
1/4 BUSHING
1/8 WALL BUSHING PRE STRESSED TO 48,000 PSI BEFORE CYCLING
0
.005
.010
BUSHING DIAMETRICAL INTERFERENCE -- INCH

FIGURE 1

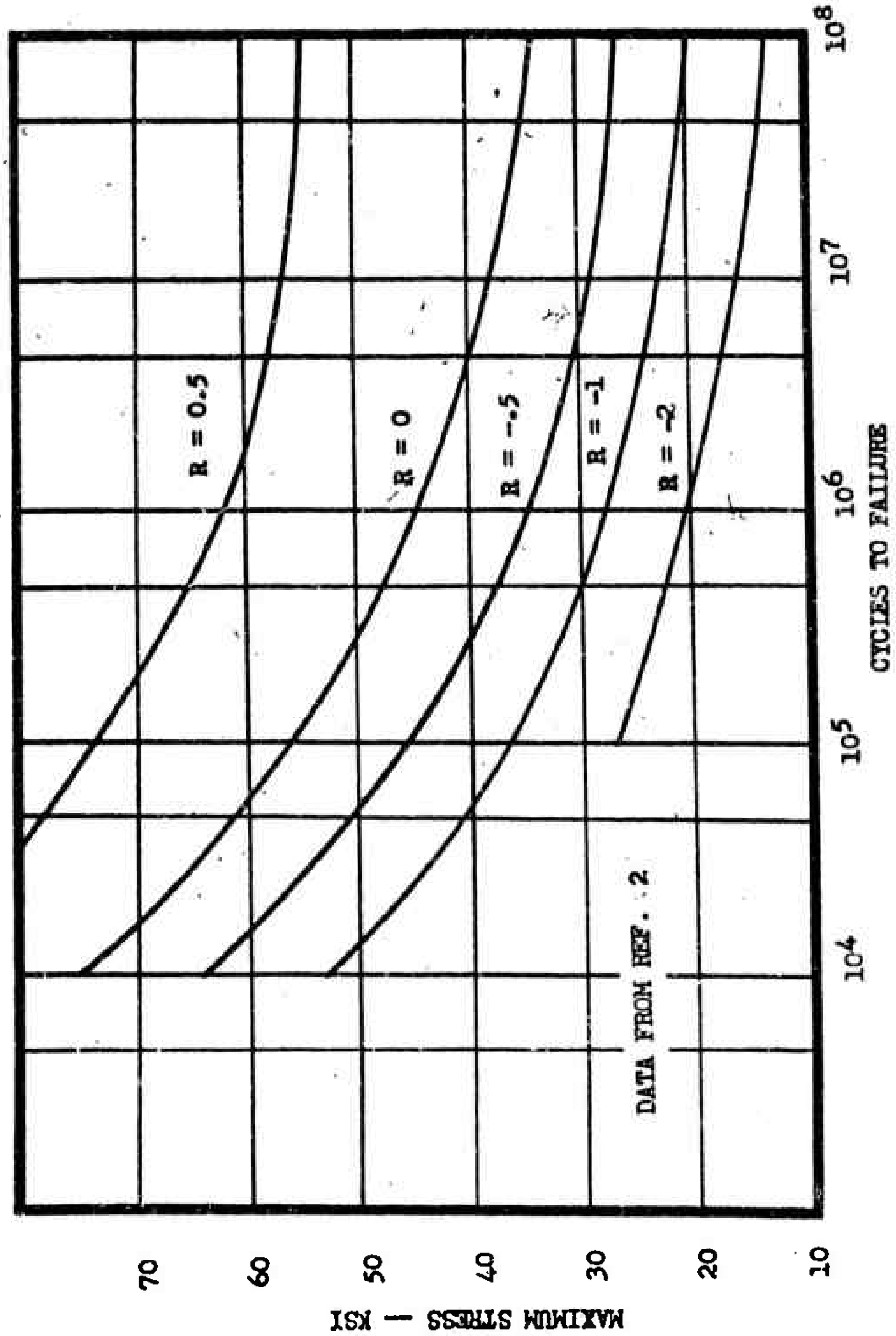

FIGURE 2 — DIRECT STRESS S N FATIGUE CURVES — 7075-T6

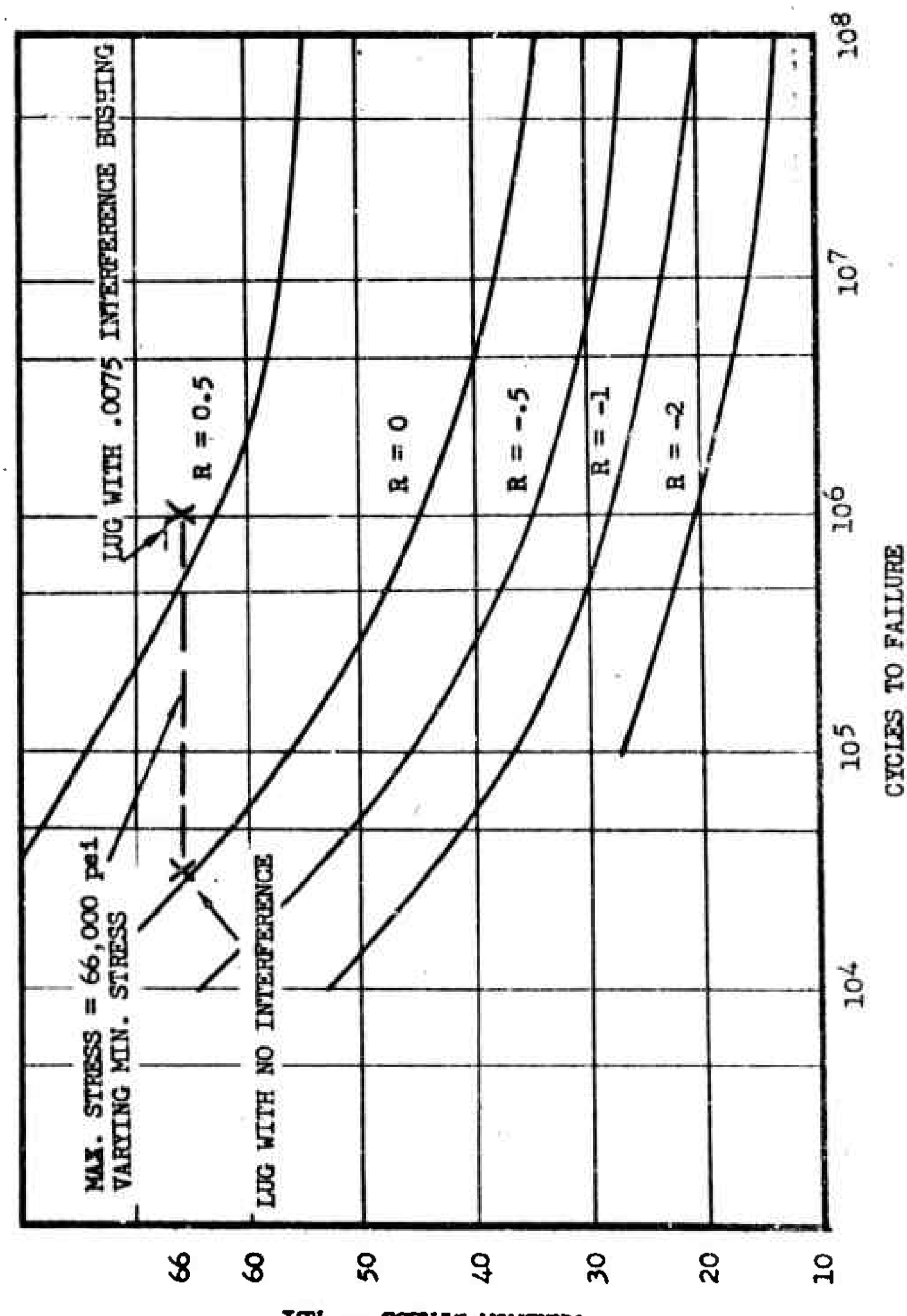

FIGURE 3 — EFFECT OF INTERFERENCE BUSHING ON FATIGUE LIFE OF LUG—K_t = 3.5

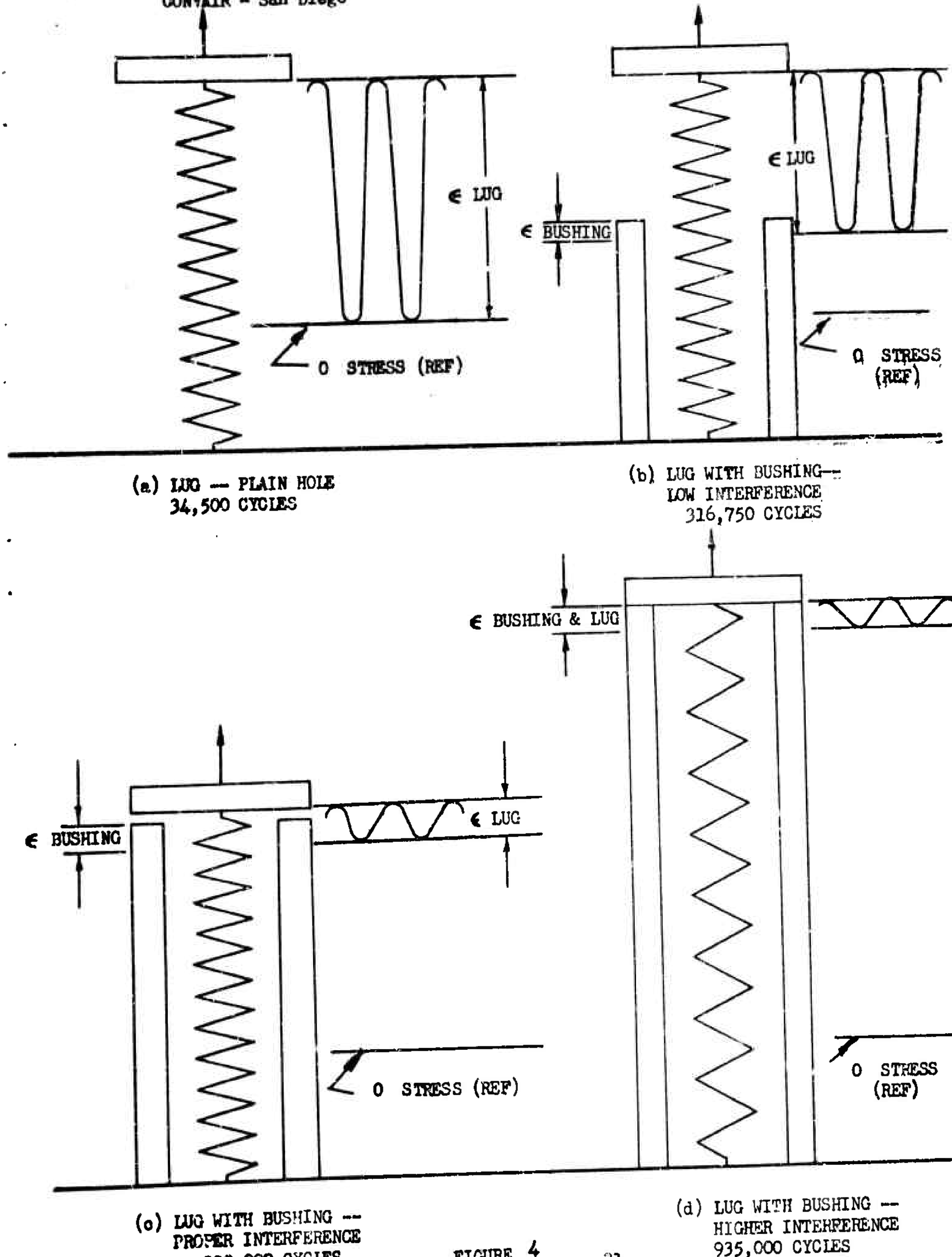
ϵ LUG
0 STRESS (REF)
(a) LUG — PLAIN HOLE
34,500 CYCLES
ϵ BUSHING
ϵ LUG
0 STRESS (REF)
(b) LUG WITH BUSHING—
LOW INTERFERENCE
316,750 CYCLES
ϵ BUSHING & LUG
ϵ BUSHING
ϵ LUG
0 STRESS (REF)
0 STRESS (REF)
(c) LUG WITH BUSHING —
PROPER INTERFERENCE
990,000 CYCLES
(d) LUG WITH BUSHING —
HIGHER INTERFERENCE
935,000 CYCLES

FIGURE 4

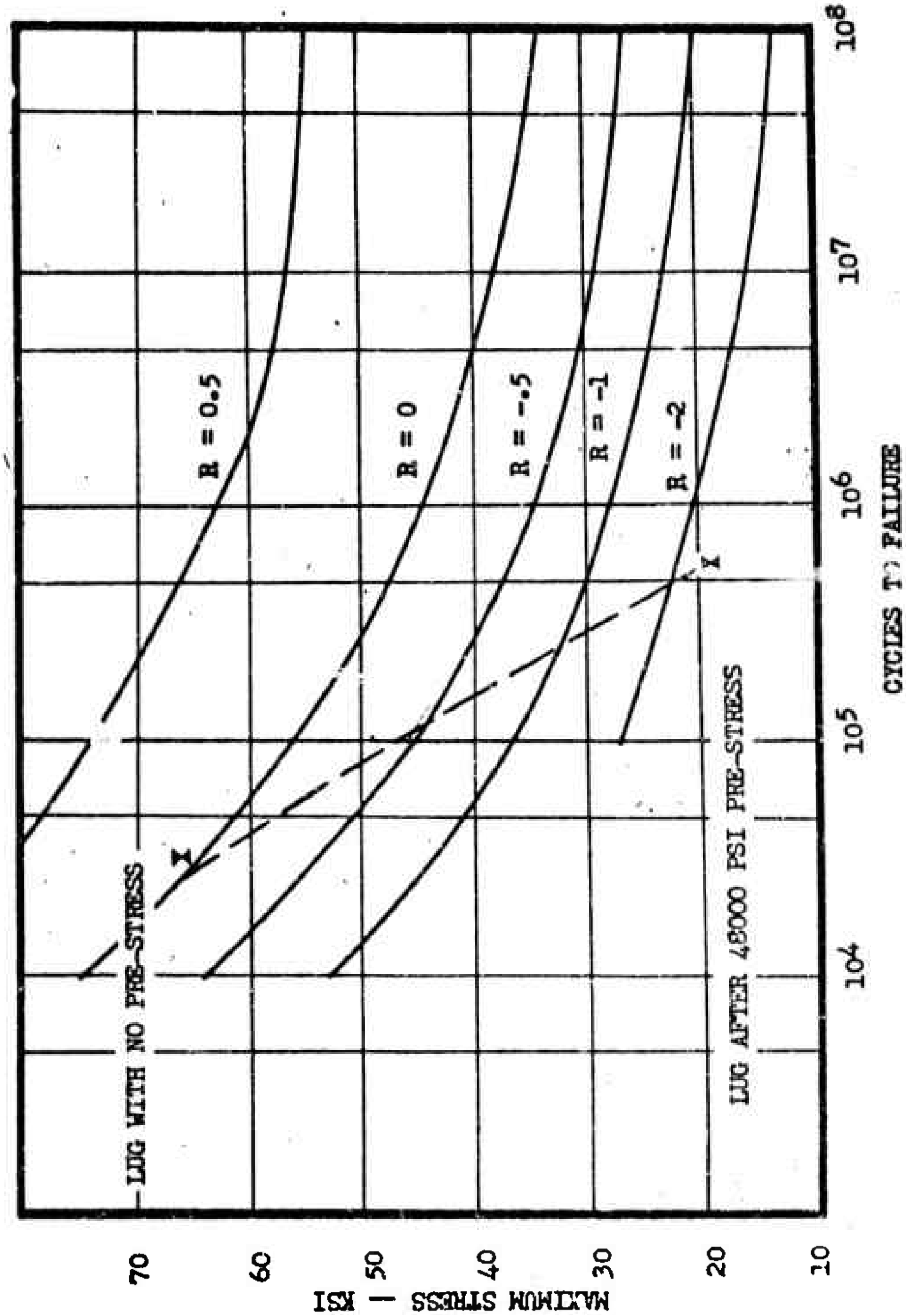

FIGURE 5 — EFFECT OF RESIDUAL STRESS ON LIFE

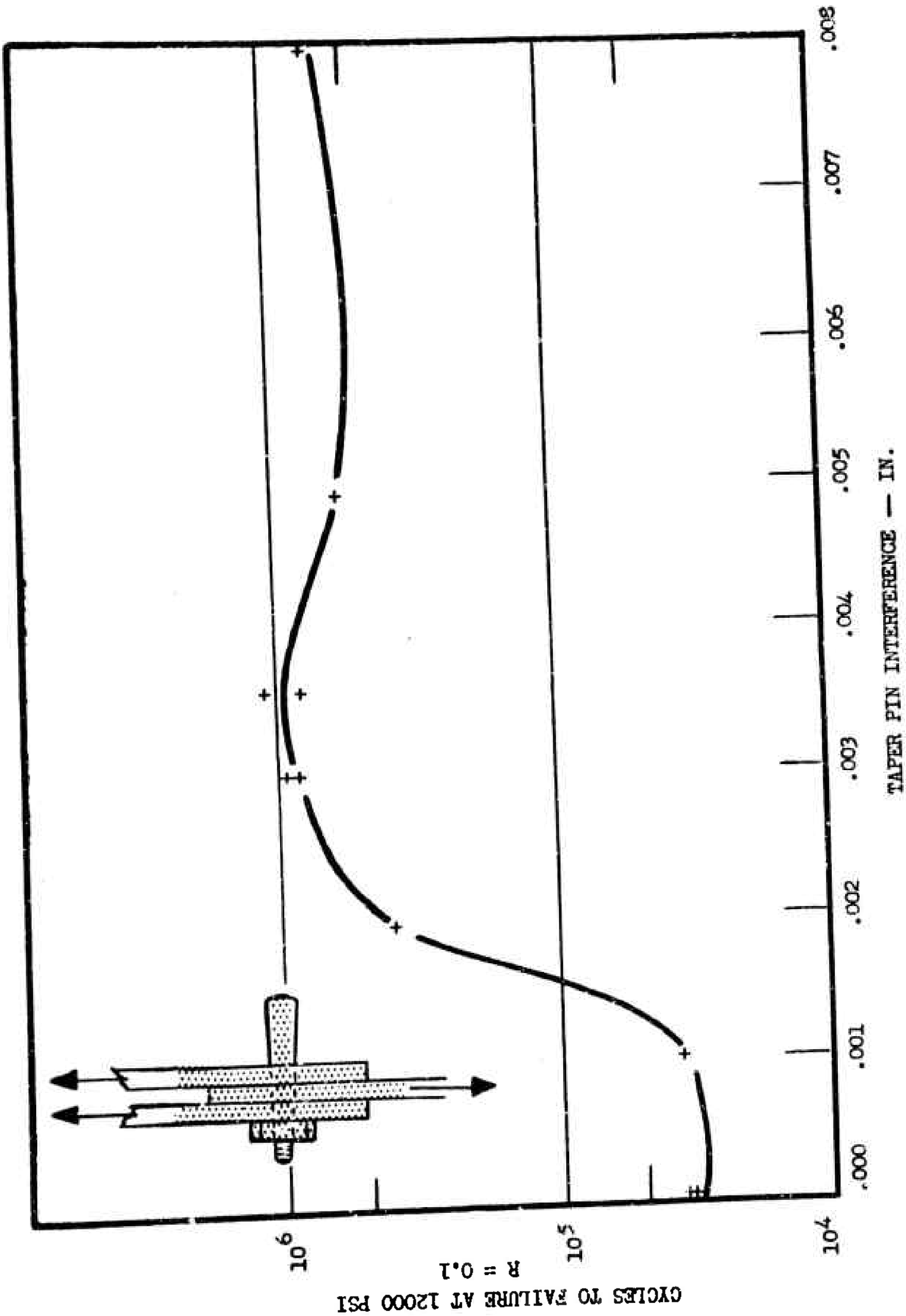
CYCLES TO FAILURE AT 12000 PSI
R = 0.1
10^4
10^5
10^6
TAPER PIN INTERFERENCE — IN.
.000
.001
.002
.003
.004
.005
.006
.007
.008

FIGURE 6

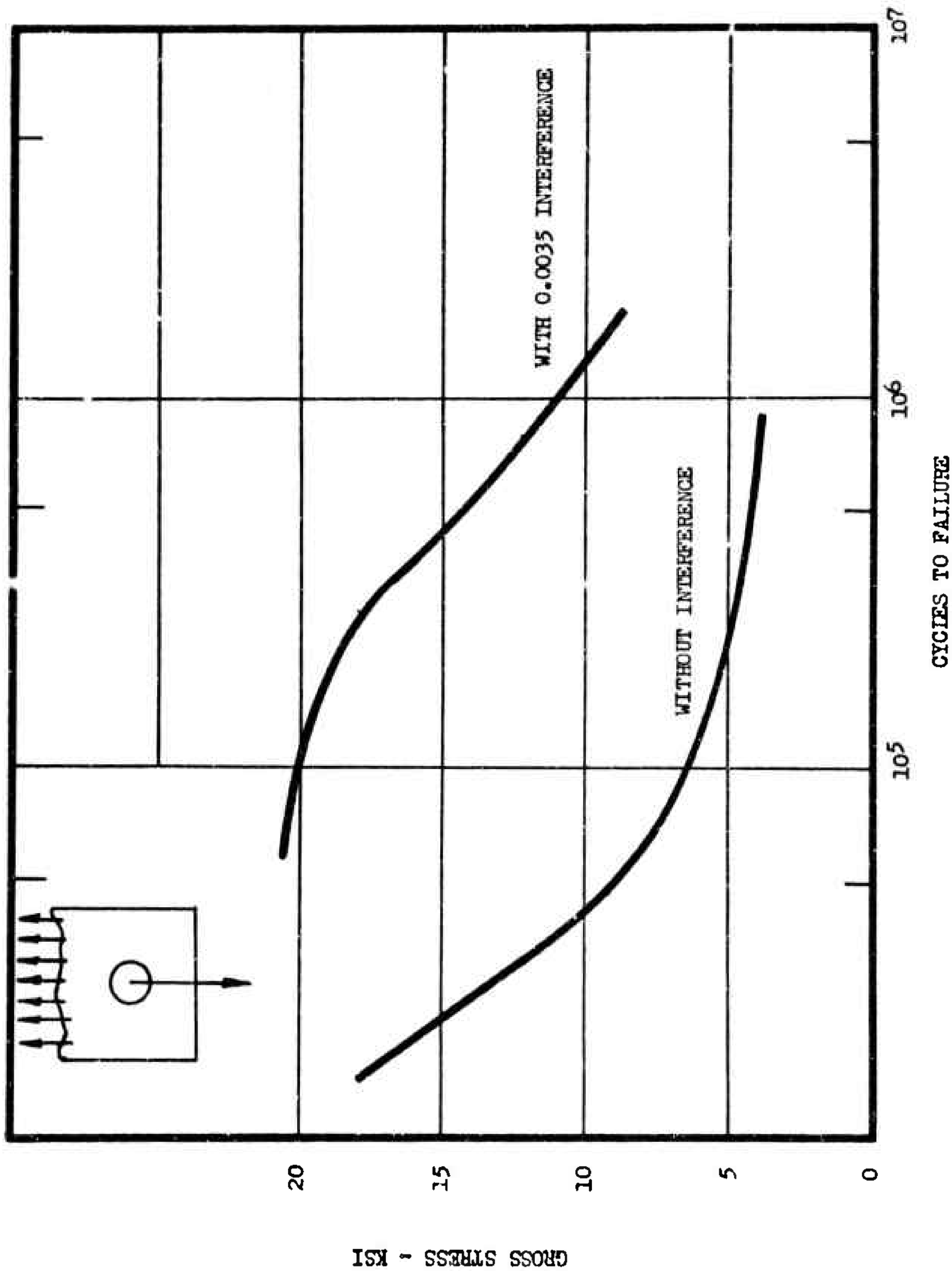

FIGURE 7 - PIN LOADED LUGS — EFFECT OF INTERFERENCE

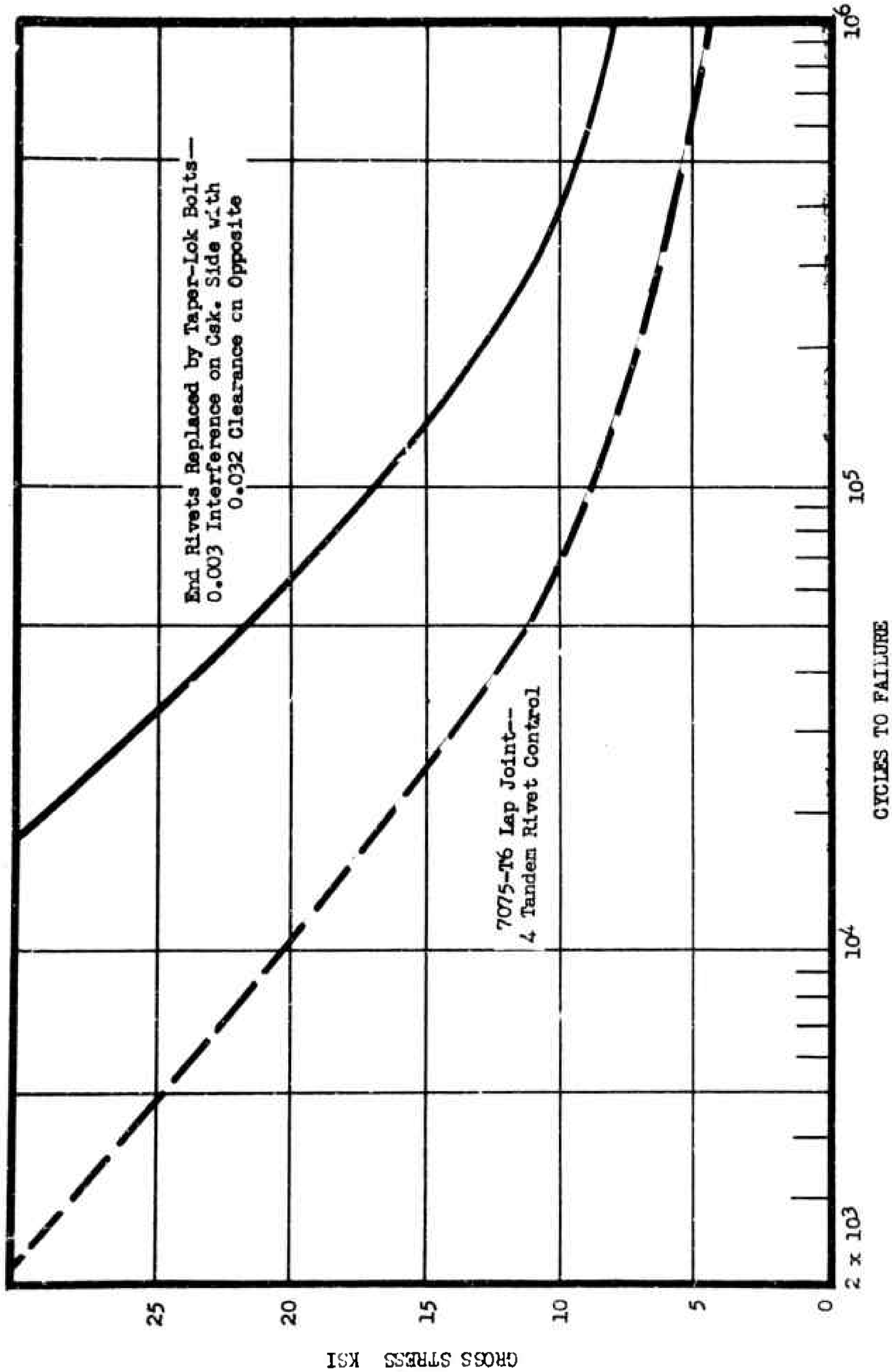

FIGURE 8 - 4 RIVET LAP JOINTS AND JOINTS WITH INTERFERENCE FIT BOLTS